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ABSTRACT

The paper describes the investigations of the R718 thermocompression refrigerating cycle with two-phase ejector as a compression device. The complex thermal and flow phenomena connected with compression and condensation inside the two-phase ejector flow field are investigated and performance characteristics are estimated. The thermodynamic processes are shown in pressure – enthalpy and temperature – entropy diagrams. Based on the described thermodynamic model, a computational procedure was generated to evaluate the performance characteristics of this R718 refrigerating system. The implementation of the proposed R718 two-phase ejector refrigerating / heat pump systems for utilization of low temperature heat and waste heat is discussed.

1. INTRODUCTION

The global environmental problems connected with ozone depletion and global warming have led to renewed interest in natural refrigerants (ammonia, carbon dioxide, water, air and hydro-carbons such as propane, butane). The theory of gas and steam ejectors is given in the fundamental publications (for example: Abramovic, 1969, Power, 1993). The theory of ejector steam jet refrigerating systems is also given in the base publications (for example: Čerepnalkovski, 1991). The fluid flow analysis and considerations presented in this work are based on the fundamental principles and publications (Loicianski, 1993), (Abramovic, 1969), (White, 2008), as well as on the numerous computational and experimental investigations published in recent years.

Theoretical and experimental investigations of the ejectors and ejector refrigerating systems working with various refrigerants are presented by Sokolov and Hershgal (1990); Eames *et al*, (1995); Cardemil and Colle (2011); Elbel and Hrnjak (2008); Huang *et al*, **(**1999); Petrenko *et al*, (2011). The optimization of the ejector flow field depends on refrigerating system operating conditions. Theoretical and experimental investigations of the two phase flow ejectors applied in the compressor refrigerating systems as devices for reduction of the throttling losses, or for second step compression in the refrigerating cycles are presented by Šarevski *et al*, (2005); Bergander *et al*, (2006, 2008); Butrymowicz *et al*, (2003, 2007); Elbel and Hrnjak, (2008a); Smierciew *et al*, (2011); Banasiak *et al*, (2011); Wang *et al*, (2009). The improvement of the COP of compressor refrigerating system working with ammonia (R717) is estimated to be about 4% (Šarevski *et al*, 2005), for t_c=40 ^{o}C, t_e=0 ^{o}C, and about 8% for HFC 134a. According to experimental investigations of the R744 refrigerating system (Banasiak *et al*, 2011) the COP improvement is about 8%. The two phase ejector efficiency, defined as a ratio between isentropic compression work from the evaporating pressure to the ejector outlet pressure and maximum possible isentropic expansion work from the condensing pressure to the evaporating pressure, (Banasiak *et al*, 2011) is maximum 0.35.

General theory of thermotransformation is used for analysis of the thermocompression systems applied in the concentrating processes (Šarevski, 2009; 2011). Optimization of the steam ejector thermocompression vacuum pump and application in oil deodorization processes is presented by Šarevski *et al*, (1999). Increment of the energy efficiency of the industrial steam-condense systems can be achieved by implementation of the ejector thermocompression (Šarevski *et al*, 2003). Energetic and exploitation characteristics of the two-phase ejector vacuum system installed in a paper machine are analyzed by Šarevski *et al*, (2004).

Calculation and analysis of sound velocity in vapor liquid two phase flow, as well as theoretical and experimental investigations of transonic flow phenomena in two phase ejectors are given by Wang and Zhang (2011); Karwacki *et al*, (2011). The possibilities for implementation of two phase ejectors in the compressor refrigeration systems and heat pumps working with natural refrigerants and estimation of the energy efficiency of these systems are presented by Šarevski (2012a).

The purpose of this paper is to describe R718 thermocompression refrigerating cycle with two-phase ejector as a compression device, to estimate performance characteristics of this cycle and to give possibilities for implementation of two phase ejectors in the refrigeration and in the heat pump systems for utilization of low temperature heat and waste heat.

2. EJECTOR MAIN CHARACTERISTICS

Compression in the ejector systems is realized by using motive fluid with high pressure. Boiler steam is used as a motive fluid in the steam jet ejector vacuum systems (Šarevski *et al*, 1999), steam jet ejector refrigerating systems (Čerepnalkovski, 1991), thermocompression heat pump systems (Šarevski *et al*, 2011) etc. Low temperature waste heat and also solar energy can be used for motive steam production (Sokolov and Hershgal, 1990; Petrenko *et al*, 2011; Cardemil and Colle, 2011). Natural refrigerants (R744, R718) are used as working media in these ejector refrigerating systems. Water pressurized by a hydraulic pump as a motive fluid is used in the two phase ejector vacuum pump systems (Šarevski *et al*, 2004; 2009) and also in the concentrator thermocompression systems (Šarevski *et al*, 2011) for compression of the water vapor from evaporating to condensing pressure. Liquid refrigerant with condensing pressure is used as a motive fluid in the two phase ejectors for pre compression of the evaporated refrigerant (Šarevski *et al*, 1996; 2005) Compression in the two phase ejector refrigerating systems is realized by using motive water with high pressure (Šarevski, 2012b).

The two phase ejector flow analysis is based on the assumption that saturated vapor-liquid mixture is in thermodynamic equilibrium state at any cross-section of the ejector, and that liquid and vapor are uniformly mixed and flow at the same velocity without inter-phase slip. In the ejector primary nozzle motion fluid accelerates and expands (1-2) (Figure 1) from the high pressure p_1 to the pressure p_2 which is lower than secondary flow suction pressure. The flow at the outlet of the primary nozzle is usually supersonic, and the nozzle profile is convergent-divergent. At primary nozzle throat cross-section the pressure is equal to the critical and the velocity is equal to the sound velocity $a=\sqrt{\partial p/\partial \rho}$. Viewing from the existing literatures there is a lack of sound velocity data in two phase flow (Wang and Zhang, 2011). The calculation of the sound velocity and the flow analyses of the two phase ejectors are complex tasks. The evaluation of the sound velocity and fluid flow analyses can be carried out by numerical methods ($a=\sqrt{(\Delta p/\Delta \rho)_{s=const}}$).

The steady-state and steady-flow equations of energy, momentum and continuity are:

- energy equation for an adiabatic process,
$$\sum M_i(h_i+c_i^2/2)=\sum M_e(h_e+c_e^2/2) \tag{1}$$
- momentum equation,
$$p_iA_i+\sum M_ic_i=p_eA_e+\sum M_ec_e \tag{2}$$
- continuity equation,
$$\sum \rho_ic_iA_i=\sum \rho_ec_eA_e \tag{3}$$

The primary nozzle outlet velocity is
$$c_2=\Psi_{pr}c_{2a}=[2(h_1-h_2)]^{1/2}=(2\Delta h_a\eta_{pr})^{1/2} \tag{4}$$
The ejector primary nozzle profile can be obtained applying the energy and continuity equations
$$M=V\rho=Ac\rho=const.;\qquad c=\sqrt{2\Delta h};\qquad A=Mv/c=Mf;\qquad f=v/c; \tag{5}$$
If isentropic change of state is assumed ($s=const.$), then dryness x, specific volume v and density ρ are
$$s=xs''+(1-x)s'=const.;\qquad x=(s-s')/(s''-s');\qquad v=xv''+(1-x)v'; \tag{6}$$
The inlet state of the water is: $p=p_1$; $t=t_1$; $x=0$; $h=h_1$; $s=s_1$. The calculated procedure for primary nozzle cross-section area is: $p=p-\Delta p \Rightarrow t \Rightarrow \Delta h \Rightarrow c \Rightarrow x \Rightarrow v \Rightarrow \rho \Rightarrow \Delta\rho \Rightarrow a \Rightarrow f \Rightarrow A$.
The primary nozzle throat cross-section $A=A_{min}$, $(f=f_{min})$, is critical cross-section, the pressure is critical pressure $p=p_{cr}$, and the velocity is equal to the sound velocity $c=a$.
The angle of converging section is 30° and the angle of diverging section is 2° for two phase ejector primary nozzle in the work presented by Banasiac *et al*, 2011. In the work presented by Karwacki *et al*, 2011, the nozzle converging section is profiled, and the angle of diverging section is 8°. In this work is suggested the converging section and the diverging section for two phase ejector primary nozzle to be profiled according to the previous given procedure, assuming constant pressure decrement gradient or constant velocity increment gradient.

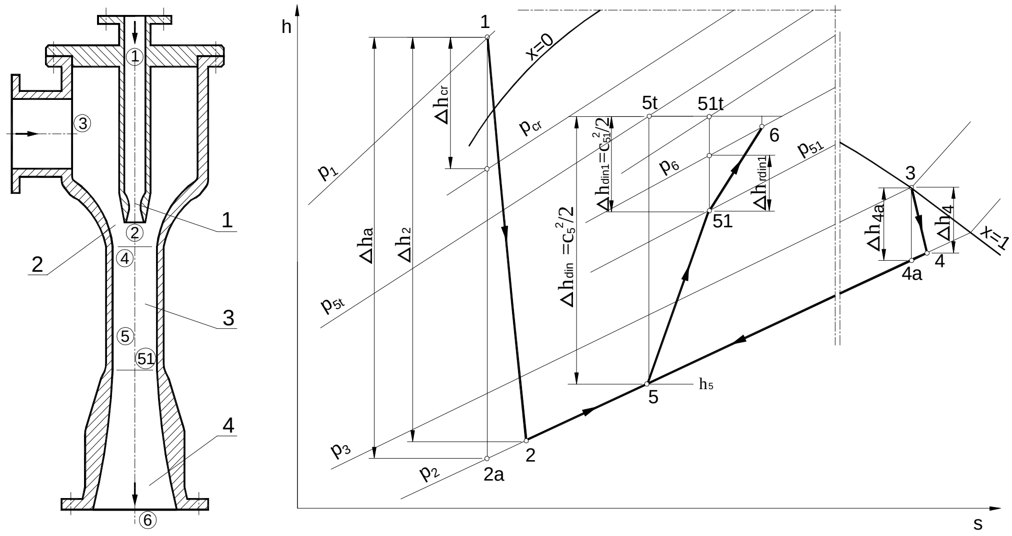


Figure 1. Scheme of an ejector and *h–s* diagram

This primary flow draws and entrains the secondary flow in the mixing chamber. The secondary flow comes through the secondary nozzle where it expands (3-4). The velocity c_4 is

$$c_4 = \Psi_{sec} c_{4a} = [2(h_3 - h_4)]^{1/2} = (2\Delta h_{4a}\eta_{sec})^{1/2} \tag{7}$$

The secondary nozzle is formed by the outside profile of the primary nozzle and inside profile of the secondary nozzle. The profiling procedure of the secondary nozzle is similar to that of the primary nozzle. The combined flows are mixed in the mixing chamber (2-5, 4-5), where appears a complex process of momentum transfer. By using the momentum equation for the mixing chamber,

$$p_2 A_2 + p_4 A_4 + M_{pr} c_2 + M_{sec} c_4 = p_5 A_5 + (M_{pr} + M_{sec}) c_5 + P_{fr} \tag{8}$$

Assuming that the cross section areas are $A_2+A_4=A_5$, pressures $p_2=p_4=p_5$ and if the friction forces P_{fr} are comprised with mixing chamber efficiency coefficient η_{mc}=0.95–0.98, the velocity of the combined flow is

$$c_5 = \eta_{mc}(c_2 m_{pr} + c_4 m_{sec});\ m_{pr} = M_{pr}/(M_{pr} + M_{sec});\ m_{sec} = M_{sec}/(M_{pr} + M_{sec}); \tag{9}$$

The main losses in the ejector occur in the mixing chamber in the process of momentum transfer. The loss of the kinetic energy or loss of the total pressure is

$$\Delta E = E_1 + E_2 - E_5 = M_{pr} c_2^2/2 + M_{sec} c_4^2/2 - (M_{pr} + M_{sec}) c_5^2/2 = 0{,}5 M_{pr} M_{sec}(c_2^2 - c_4^2)/(M_{pr} + M_{sec})$$

$$\delta_e = \Delta E / E_1 = m_{sec}(c_2^2 - c_4^2)/c_2^2 = (1 - m_{pr})(c_2^2 - c_4^2)/c_2^2 \tag{10}$$

However, in these ejector systems primary mass flow rate is much larger than secondary mass flow rate ($m_{pr}>>m_{sec}$). Therefore this loss of total pressure is negligible. This is significant condition for achievement of high energy efficiency by these thermocompression ejector systems.

The enthalpy of the combined flow is obtained by using the energy equation of the mixing chamber,

$$h_5 = m_{pr} h_2 + m_{sec} h_4 + m_{pr} c_2^2/2 + m_{sec} c_4^2/2 - c_5^2/2 \tag{11}$$

The optimal length of the mixing chamber depends on the ejector operating conditions. According to theoretical and experimental investigations and experience the optimum length is L_{mc}=(8–10)D_{mc}. The compression of the fluid is achieved as the combine stream flows through the diffuser. The kinetic energy $\Delta h_{din}=c_5^2/2$ in the diffuser is transformed to enthalpy rise, expressed by rise of the pressure, according to the Law of Energy Conservation. The combined flow at the mixing chamber outlet often is supersonic. If the velocity of the combined flow is supersonic then a normal shock wave occurs. The shock wave is a process where sudden change in the flow space appears, the velocity suddenly falls from supersonic to subsonic and the pressure rises. In two phase flow this complex process is accompanied by mass transfer from one phase to the other. Mach number of the supersonic flow, upstream of the shock wave is $\lambda_1=c_5/a_5>1$. Mach number downstream of the shock wave is $\lambda_2=c_{51}/a_{51}<1$. Across the shock wave $\lambda_1\lambda_2=1$. The sound velocity a_5 and a_{51} can be estimated by numerical method. Using the conditional isentropic exponent (Karwacki *et al*, 2011) and/or numerical estimation of the conditional isentropic exponent and according to the gas dynamic theory the pressure rise across the shock wave can be approximately estimated

$$\frac{p_{51}}{p_5} = \frac{\lambda_1^2 - (\kappa-1)/(\kappa+1)}{1 - (\kappa-1)\lambda_1^2/(\kappa+1)} \tag{12}$$

In the shock wave partially the compression is realized. However, the shock wave is thermodynamic irreversible process, with entropy rise. Additional compression is realized in the subsonic diffuser. According to wide range of publications about subsonic diffuser hydraulic losses, the values of diffuser efficiency η_d are from 0.60 up to 0.80, depending on shape and operating conditions. According to the resent experimental investigation of the two phase ejectors (Elbel and Hrnjak, 2008a; Banasiak *et al*, 2011) the diffuser angle of divergence is 3^o–5^o. The optimal diffuser angle of divergence for these systems is expected to be lower because the amount of liquid in the mixture is much larger.

3. R718 REFRIGERATING SYSTEM WITH TWO PHASE EJECTOR

A scheme of a single stage R718 refrigerating system with two phase ejector as a compressor device, and *T–s* and *p–h* diagrams of the processes are given in Figure 2. The vapor from the evaporator comes into the two phase ejector, where complex thermal and flow phenomena connected with compression, and condensation inside the two-phase ejector flow field appears. This system could be optimally used as a high temperature compressor / condenser unit in multi stage refrigeration systems, for example in R718 centrifugal refrigeration systems (Šarevski, 2012c, 2012d), or in R744 refrigeration systems, to avoid super critical cycle parameters.

The evaporator and condenser are with direct evaporation and condensation, without surface heat exchangers. Cooling towers can be direct (Figure 2a) or indirect (Figure 2b). In the direct cooling towers (Figure 2a) water (R718) is exposed directly to the atmospheric contamination with non-condensable gases, solid particles and liquids. That causes additional costs associated with degasifying and cleaning. The vacuum in the system is maintained by a small two phase water ejector vacuum pump, which played a supporting role, working intermittently for pumping a small amount of non-condensable gases. In the indirect cooling towers (Figure 2b) water (R718) circulates through cooling tower heat exchanger, where heat transfer is still enhanced by wetting the outside of the heat exchanger and utilizing evaporative cooling effect.

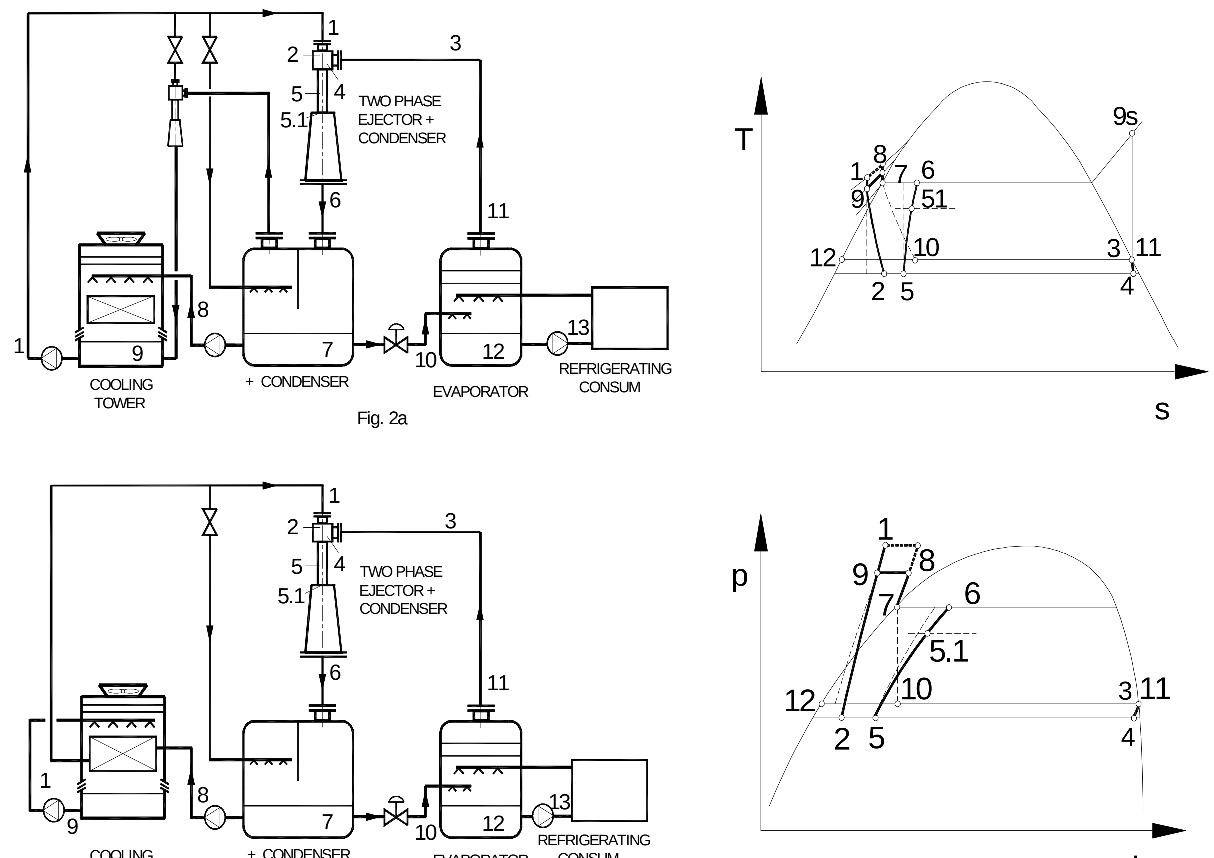


Figure 2. Scheme of a single stage R718 refrigerating system with two phase ejector and *T–s* and *p–h* diagrams of the processes

The COP of single stage R718 refrigerating system with two phase ejector is

$$COP = q_e \,/\, l_e \tag{13}$$

where are: $q_e = h_{11} - h_{10}$ – specific evaporating heat;
$l_e = (m_{pr}/m_{sec})(\Delta p/(\rho_l \eta_{pump}))$ – two phase ejector specific equivalent compression work;
$P_{pump} = M_{pr}\Delta h_{pr} / \eta_{pump} = M_{pr}\Delta p_{pr} / (\rho_l \eta_{pump})$ – pump power consumption;
M_{pr} – pump flow rate (primary flow); Δp_{pr} – pump pressure rise; ρ_l – water mean density;
η_{pump} – pump efficiency;

A scheme of a two stage R718 refrigerating system with two-phase ejectors is given in Figure 3. The COP of two-stage R718 refrigerating system with two-phase ejectors is

$$COP = q_e \,/\, (l_{e1} + k\, l_{e2}) \tag{14}$$

where l_{e1} and l_{e2} are two-phase ejector specific equivalent compression work for first and second stage; and k is coefficient of flow rate $k \approx (q_e + l_{e1})/\, q_e$.
Numerical experiments have been realized for various evaporating and condensing temperatures and various pump characteristics (M_{pr}, Δp_{pr}, η_{pump}), according to the previously explained calculating procedure. The sound velocity is calculated numerically, as well as the isentropic exponent and the profile of the primary nozzle. The calculations are performed for: pump efficiency η_{pump}=0.8; primary nozzle efficiency η_{pr}=0.9–0.95; secondary nozzle efficiency η_{sec}=0.9–0.95; mixing chamber mechanical efficiency coefficient η_{mc}=0.95–0.98; diffuser efficiency η_d =0.60–0.80. The estimations of the characteristics of the two phase ejector refrigerating systems show the following main features:
- flow at the outlet of the primary nozzle is supersonic, the nozzle profile is convergent-divergent;
- primary mass flow rate is much larger than secondary mass flow rate ($m_{pr} >> m_{sec}$), the loss of total pressure in the mixing chamber is negligible;
- compression is realized in the shock wave, the shock wave is thermodynamic irreversible process with loss and entropy rise;
- additional compression is realized in the subsonic diffuser;
- total efficiency of the two phase ejector (η_e) defined as a ratio between isentropic compression power from the evaporating pressure (3, Figure 2) to the condensing pressure (point 9s) and pump power consumption is obtained to be in the range η_e=0.30–0.53, according to the results of numerical experiments and previously explained calculating procedure. The efficiency of two phase ejector (η_e) depends on the efficiency of the ejector flow parts, hydraulic pump characteristics, temperature lift $\Delta t=(t_c-t_e)$ (Δt=(15–5) ^{o}C), and subcooling water temperature at the ejector primary nozzle $\Delta t_1=(t_c-t_1)$ (Δt_1=(0–7) ^{o}C).
- coefficient of performance *COP* of the refrigeration stage with two-phase ejector, according to the numerical experiments is estimated to be in range:
    COP = (22.5–25.5) for temperature lift $\Delta t=(t_c-t_e)=5^o$C; (η_e=0.47–0.53),
    COP = (14.4–16.6) for temperature lift $\Delta t=(t_c-t_e)=8^o$C; (η_e=0.42–0.51),
    COP = (9.7 –11.8) for temperature lift $\Delta t=(t_c-t_e)=11^o$C; (η_e=0.34–0.48),
    COP = (7.0 – 8.6) for temperature lift $\Delta t=(t_c-t_e)=15^o$C; (η_e=0.30–0.45),
depending on the efficiency of the ejector flow parts and subcooling water temperature at the ejector primary nozzle $\Delta t_1=(t_c-t_1)$ (Δt_1=(0–7) ^{o}C).

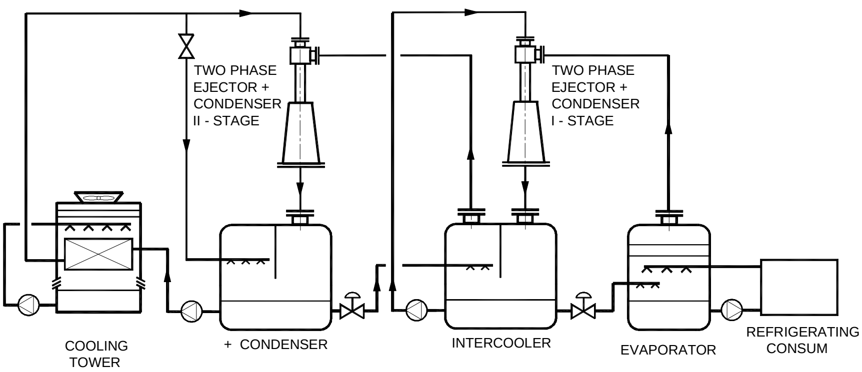


Figure 3. Scheme of a two stage R718 refrigerating system with two phase ejectors

The COP of two-stage R718 refrigerating system with two-phase ejectors (Figure 3) for air conditioning application, with temperature conditions: t_e=10 °C (cooling water 10/15 °C) and t_c=35 °C (condensing water 35/30 °C), according to the numerical experiments is estimated to be in range of *COP* = 4.0–4.7.

The task of optimization of the ejector flow field needs further theoretical (CFD simulations; cycle calculating procedures; thermal and flow analyses of the two phase processes) and experimental investigations.

4. EJECTOR THERMOCOMPRESSION SYSTEMS APPLIED IN CONCENTRATORS

The waste water vapor from the concentrating process can be directly compressed to a higher pressure level, so the waste heat will be transformed to a higher temperature level convenient for usage in the thermal concentrating process. The industrial concentrators have wide application in the technological processes of the chemical and pharmaceutical industries, in the dairy, beer and sugar industries, in the industrial plants for productions of fruit, tomato and grape concentrates etc. Large consumption of energy is a common characteristic of the industrial concentrators. Decreasing of energy consumption can be achieved by multi stage concentrating and by using thermocompression heat pumps.

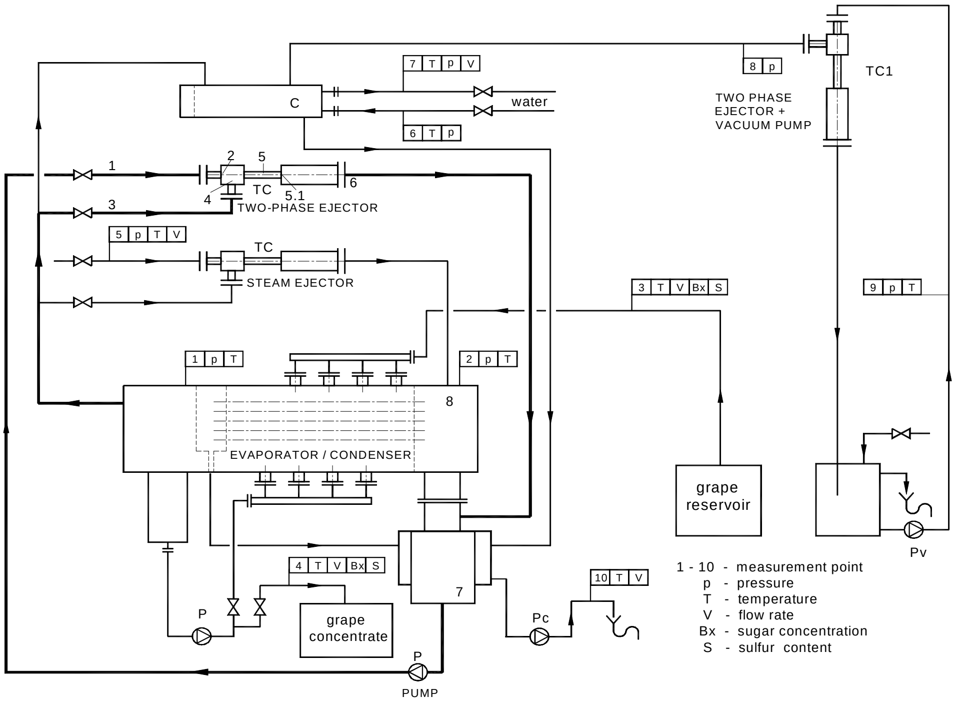
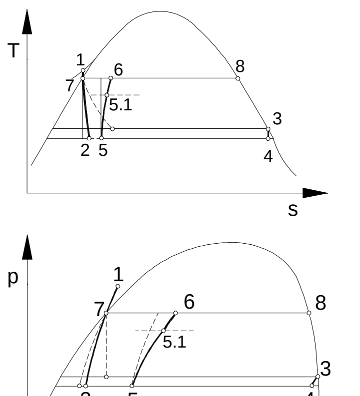


Figure 4. Scheme of an experimental grape concentrator



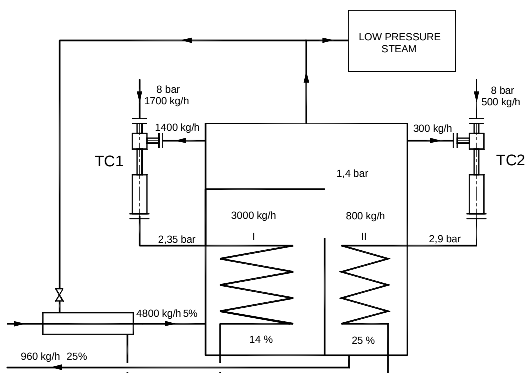


Figure 5. Diagrams of processes in two phase ejector **Figure 6**. Scheme of a NaOH concentrator

The theoretical and experimental investigations on an experimental grape vacuum concentrator (Figure 4) and on an industrial NaOH concentrator (Figure 6) show that significant increment of the energy efficiency can be achieved by optimal application of thermocompression in thermal concentrating processes (Šarevski, 2011, 2012b). The steam ejector thermocompression applied in a grape vacuum concentrator can achieve high thermotransforming coefficient (Ψ_e = 3.1–5.2), which provide significant increment of the concentrator energy efficiency. Because of the increment of the NaOH solution boiling temperature, lower thermotransforming coefficient can be obtained in a NaOH concentrator (Ψ_e =1.6–1.8).

The experimental grape vacuum concentrator was equipped with water two-phase ejector and mechanical centrifugal multi stage pump (Figure 4). Compression in the two phase ejector systems is realized by using motion water with high pressure. Water pressurized by the hydraulic pump as a motion fluid is used in the two phase ejector system for compression of the evaporated water from evaporating to condensing pressure. Pressure–enthalpy (*p-h*) and temperature–entropy (*T-s*) diagrams of the processes are given in Figure 5.

The thermotransforming coefficient Ψ_e , or coefficient of performance COP_h is

$$\Psi_e = COP_h = (M_{sec}\, q_e + P_{pump}) / P_{pump} = q_e / l + 1 \tag{15}$$

where: $q_e = h'' - h'$ – specific evaporating heat;
$l = (m_{pr}/m_{sec})(\Delta p/(\rho_l \eta_{pump}))$ – two phase ejector specific equivalent compression work;
$P_{pump} = M_{pr}\Delta h_{pr} / \eta_{pump} = M_{pr}\Delta p_{pr} / (\rho_l \eta_{pump})$ – pump power consumption;
M_{pr} – pump flow rate (primary flow); Δp_{pr} – pump pressure rise; ρ_l – water mean density;
η_{pump} – pump efficiency;

Numerical experiments have been realized for various pump characteristics (M_{pr}, Δp_{pr}, η_{pump}), and for various evaporating temperatures, according to the previously explained calculating procedure. Total efficiency of the two phase ejector is estimated to be in the range (η_e=0.34–0.53), depending on the efficiency of the ejector flow parts, hydraulic pump characteristics (η_{pump}=0.8) and thermo transforming temperature lift $\Delta t=(t_c - t_e)$ (Δt=5–11 ^oC). The thermo transforming coefficient Ψ_e , or coefficient of performance COP_h according to the numerical experiments is estimated to be in range:

$\Psi_e = COP_h$ = (23.5–26.6) for temperature lift $\Delta t=(t_c - t_e)$=5^{o}C; (η_e=0.47–0.53),
$\Psi_e = COP_h$ = (14.4–16.6) for temperature lift $\Delta t=(t_c - t_e)$=8^{o}C; (η_e=0.42–0.46),
$\Psi_e = COP_h$ = (9.7–11.8) for temperature lift $\Delta t=(t_c - t_e)$=11^{o}C; (η_e=0.34–0.40),

depending on the efficiency of the ejector flow parts (η_{pr}=0.9–0.95; η_{sec}=0.9–0.95; η_{mc}=0.95–0.98; η_d=0.60–0.80).

Preliminary experimental investigations were conducted as a part of an investigated project (Šarevski *et al*, 2005a). The results of the experiments were successful. The achieved pressure lift (and corresponding temperature lift) was satisfied. There were some problems with pump performance characteristics and with pump`s hermetic properties. The experiments were performed for one configuration of the ejector flow field, and with a given hydraulic pump. It is necessary to conduct further experimental investigations to obtain optimal two phase ejector flow field, as well as optimal performance characteristics of the motion pump. The two phase ejector calculating procedure will be improved with obtained experimental data.

5. CONCLUSIONS

R718 thermocompression refrigerating cycle with two-phase ejector as a compression device is proposed and described. The calculating procedure for estimation of the two phase ejector main parameters presented in this paper and analyses of the thermal, flow and performance characteristics of the ejectors show that the implementation of the two phase ejectors in the refrigeration systems and heat pumps working with R718 can be successful, resulting on simplification of the units, reduction of their size and cost, as well as on the possibilities of utilization of low temperature heat and waste heat.

The investigations refrigeration systems with two phase ejectors as a compression device show that COP is reasonably high, although the efficiency of two phase ejector is relatively low. Direct evaporation and condensation in these R718 refrigeration systems are the reasons for achievement of reasonably high COPs. Two-stage R718 refrigeration system with two phase ejectors is proposed for temperature conditions for air conditioning application. For t_e=10 ^{o}C (cooling water 10/15 ^{o}C) and t_c=35 ^{o}C (condensing water 35/30 ^{o}C) the COP of this system is estimated to be in range (4.0–4.7).

R718 refrigeration stage with two phase ejector could be optimally used as a high temperature compressor / condenser unit in multi stage refrigeration systems, for example in R718 centrifugal refrigeration systems, or in R744 refrigeration systems, to avoid super critical parameters of the cycle.

According to the results of numerical experiments and calculating procedure the total efficiency of two phase ejector (η_e) defined as a ratio between isentropic compression power from the evaporating pressure to the condensing pressure and pump power consumption is obtained to be in the range η_e=0.30–0.53. The efficiency of depends on the efficiency of the ejector flow parts, hydraulic pump characteristics, temperature lift $\Delta t=(t_c-t_e)$ (Δt=(15–5) ^{o}C), and subcooling water temperature at the ejector primary nozzle $\Delta t_1=(t_c-t_1)$ (Δt_1=(0–7) ^{o}C). Hydraulic losses in the compression process of transonic two phase flow have main influence on the efficiency. Compression is realized in shock wave which is thermodynamic irreversible process with loss and entropy rise.

Two phase ejector is proposed for thermocompression in the concentrating plants. Water pressurized by the hydraulic pump as a motive fluid is used in this two phase ejector thermocompression system for compression of the evaporated water from evaporating to condensing pressure. The investigations show achievement of high thermotransforming coefficients (Ψ_e=(23.5–26.6) for temperature lift $\Delta t=(t_c-t_e)=5^o$C; Ψ_e = (9.7–11.8) for temperature lift $\Delta t=(t_c-t_e)=11^o$C). It is necessary to conduct further experimental investigations to obtain optimal two phase ejector flow field, as well as optimal performance characteristics of the motion pump.

The task of optimization of the ejector flow field needs further theoretical (CFD simulations; cycle calculating procedures; thermal and flow analyses of the two phase processes) and experimental investigations.

NOMENCLATURE

a	Speed of sound (m s^{-1})
A	Flow cross-section (m^2)
c	Velocity (m s^{-1})
c_l	Water specific heat capacity (J kg^{-1} K^{-1})
COP	Coefficient of performance
D	Diameter (m)
h	Enthalpy (J kg^{-1})
l	Compression work (J kg^{-1})
M	Mass flow rate (kg s^{-1})
m	Relative mass flow rate
P_{pump}	Pump power consumption (W)
P_{fr}	Friction forces (N)
p	Pressure (Pa, bar)
q_e	Specific refrigerating effect (J kg^{-1})
s	Specific entropy (J kg^{-1} K^{-1})
T, t	Temperature (K, ^{o}C)
v	Specific volume (m^3 kg^{-1})
x	Dryness (quality)

Greek letters

Δp_{pr}	Pump pressure rise (Pa)
Δt	Temperature difference (^{o}C, K)
η	Efficiency
κ	Isentropic exponent
ρ_l	Water mean density
Ψ	Velocity coefficient
Ψ_e	Thermotransforming coefficient

Subscripts

a	Isentropic	pr	Primary
c	Condensation	sec	Secondary
cr	Critical	mc	Mixing chamber
e	Evaporation		
e	Exit	1–12; 5.1 states in	
i	Inlet	Figure 1 – Figure 4	
$\boldsymbol{e}$	Ejector		

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